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BOILER SAFETY VALVES.

By J. F. POTTINGER.

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May/51

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ILLUSTRATIONS.

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INTRODUCTION.

A steam boiler has been referred to as consisting essentially of two primary parts, the boiler itself and the pressure safety valve. This description may savour of over simplicity, but undoubtedly the importance of the safety valve as an integral part of steam generating plant should never be underrated. The disastrous damage which may result from safety valve failure needs no detailed description here.

Since the first widespread use of steam for power during the early part of the last century the safety valve has passed through many stages of development and refinement and considerable research has been conducted by leading safety valve specialists to attain the utmost technical efficiency. In this pamphlet it is hoped to give readers some idea of the many problems involved, together with details concerning modern trends in design particularly in connection with the exacting conditions met with in high pressure and high temperature steam work.

The basic function of a safety valve is to prevent the steam pressure in a boiler exceeding a predetermined maximum by automatically discharging the steam as soon as this maximum pressure is reached. Moreover, besides operating at the set pressure, safety valves must be capable of discharging the full evaporative capacity of the boiler, otherwise the possibility of continued pressure build up will remain despite perfect mechanical functioning on the part of each valve. This question of discharge capacity is very important and, as will be seen later, it has had considerable bearing on the evolution of the modern high duty safety valve.

Safety valves may be classified basically into three distinct groups according to the method employed for valve loading, viz. :— (*a*) weight loaded ; (*b*) spring loaded ; (*c*) pressure loaded. Valves in group (*a*) may be again subdivided into (i) deadweight and (ii) lever and weight types ; those loaded by direct steam pressure (group (*c*)) are generally of the relay type and under that heading will be described later in the pamphlet.

Deadweight Safety Valves.

Perhaps the most elementary type of safety valve is that in which the valve is loaded by the direct application of a weight or combination of weights above the valve. Such a valve is known as a deadweight safety valve and a typical pattern is illustrated in Fig. 1.

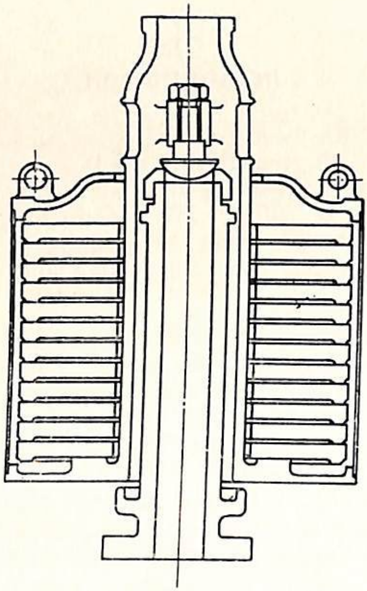

Fig. 1—Deadweight Safety Valve.

In the design shown the weights consist of circular flat cast-iron discs resting upon a supporting carriage bolted to the top side of the valve disc, the carriage being so constructed that the centre of gravity of the weights lies below the level of the valve which is thus in equilibrium. A feature is the spherical seating face of the valve; this ensures a steam tight joint regardless of slight misalignment when the valve reseats. This form of valve seat is not, however, altogether desirable and is used principally because the heavy pendulous effect of the weights renders the more orthodox wing guided valve an impracticable proposition.

The deadweight valve is probably the most certain type of safety valve and actually gives quite a satisfactory performance during operation, but it suffers from certain disadvantages which entirely preclude its employment on many types of boilers. One great drawback is its unsuitability for use on any boilers where extensive vibration and movement are experienced, as exemplified

in locomotive and marine work where its employment was discarded almost from the start. A second disadvantage arose as higher pressures became the fashion in the steam industry. This increase in working pressures brought with it the need for heavier valve loadings which, in turn, necessitated heavier and bulkier safety valves. For example, consider the case of a deadweight safety valve having a valve diameter of 3″ and designed to blow off at a pressure of 50 lbs./sq. in. Simple mathematics will show that in order to overcome the steam pressure acting on the underside of the valve and thus to maintain steam lightness up to 50 lb./sq. in., the valve must be loaded by a weight of 354 lbs. If, however, this 3″ diameter valve was to be designed to blow off at, say, 350 lb./sq. in. a load of 2,474 lbs. would be necessary, whilst at much higher pressures the loading becomes fantastic. Moreover, together with the rise in working pressures, there has been a marked increase in the evaporative capacity of boilers, particularly since the introduction of the water tube boiler, and this has brought the need for safety valves possessing large discharge areas. Now the necessary loading for a safety valve varies proportionately to the square of the valve diameter and so any increase in valve diameter, made in order to obtain a larger discharge area, may be accompanied by a prohibitive rise in valve loading.

From these considerations it will be readily apparent that the deadweight safety valve has a very limited range of application being mainly used for low pressure land boilers of the Cornish or Lancashire types.

Lever and Weight Safety Valves.

Closely allied to the deadweight safety valve is the lever and weight pattern. As will be seen from the diagrammatic representation in Fig. 2, the valve is loaded by means of a weighted

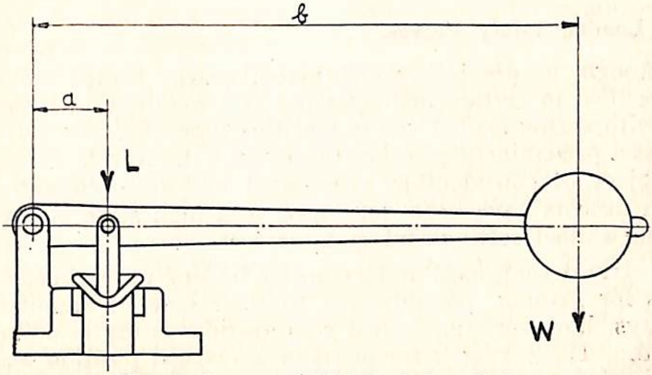

Fig. 2—Lever and Weight Safety Valve.

lever which is pivoted about a fulcrum situated close to the valve, the actual value of the valve load 'L' being dependent upon the lever ratios 'a' and 'b', viz. :—

$$L = W \frac{b}{a}$$

This compares with the expression L = W for a deadweight safety valve and, since the lever ratio b/a is usually about 8, it follows that the loading weight 'W' is something like $\frac{1}{8}$ of the weight necessary for a corresponding deadweight valve. Consequently lever and weight safety valves do not suffer to such an extent the pressure limitations experienced by deadweight valves and are occasionally used for pressures up to 600 lb./sq. in. Their employment is, in fact, still stipulated by a number of foreign countries, notably Poland, Russia and the Scandinavian States, for low and medium pressure stationary boilers but manufacturers in this country generally try to persuade the authorities concerned (often successfully) to accept the more standard spring loaded valves.

A characteristic disadvantage possessed by lever loaded safety valves might be mentioned here. Due to the necessity of at least two pivoting points (at the fulcrum and at the valve spindle) these valves are liable to suffer, to a much greater extent than other types, from the effects of friction which in bad cases may cause excessive feathering during blowdown. The fulcrum is usually in the form of a steel pin lying in a phosphor bronze bushing but friction may be further reduced by the provision of knife edge fulcrum points.

In order to prevent unauthorized tampering or inadvertent misplacement the weight is usually locked securely to the lever by a dowel pin and padlock or similar means. In certain cases, however, the covering specification calls for a lock-up case which is designed to completely encase the weight and lever.

Spring Loaded Safety Valves.

Although, as previously mentioned, weight loaded valves are still specified in certain instances the vast majority of boilers are fitted with spring loaded valves and this type of safety valve now occupies a pre-eminent position throughout the world. It has been the subject of considerable experiment and research and many modern designs have been developed to a high state of precision requiring skilled setting and maintenance.

An early design, used on locomotive boilers, utilised coach type springs for loading, possibly due to their ease of manufacture ; nowadays, however, the helical compression spring is universally adopted. The spring is mounted in a vertical position and acts immediately above the valve, initial compression and setting being

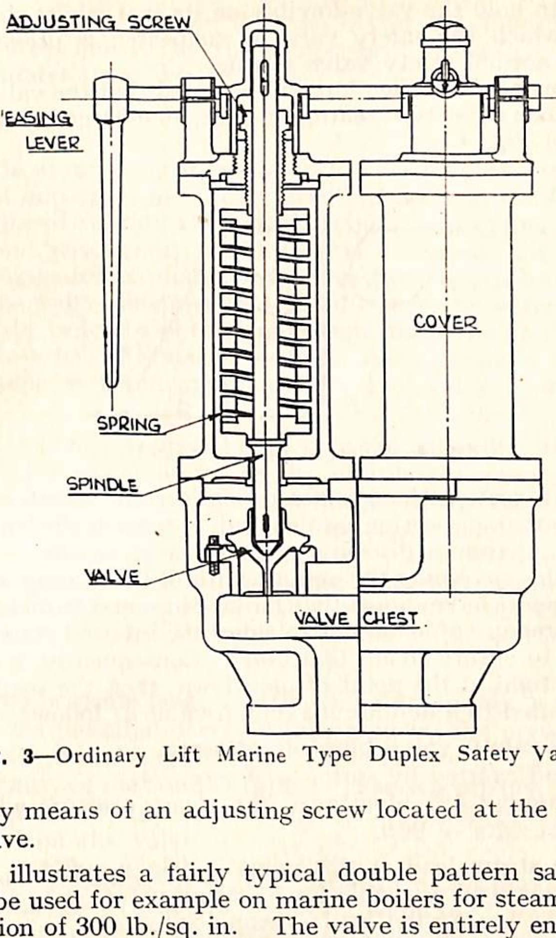

Fig. 3—Ordinary Lift Marine Type Duplex Safety Valve.

effected by means of an adjusting screw located at the top of the safety valve.

Fig. 3 illustrates a fairly typical double pattern safety valve as might be used for example on marine boilers for steam pressures in the region of 300 lb./sq. in. The valve is entirely enclosed and discharged steam is led away to atmosphere via the central outlet chamber. Ventilation ports are sometimes provided in the valve casing to enable circulating air to cool the spring, whilst on equivalent patterns intended chiefly for land boilers the spring may be left completely exposed to atmosphere. Locomotive safety valves often have a muffle incorporated in the outlet chamber.

Easing or testing levers, the provision of which is generally compulsory, enable the valve to be lifted manually off its seat under working conditions, thus ensuring the maintenance of complete mechanical efficiency which may be to some extent impaired by long periods of inactivity. Arrangements for valve "gagging" are often incorporated. This may take the form of a clamp which

is applied to hold the valve forcibly on its seat whilst the pressure vessel to which the safety valve is connected is pressure tested above the normal safety valve setting.

Referring again to Fig. 3, it will be noted that the valve element illustrated has a flat type seating surface, an enlarged view of which is shown in Fig. 4.

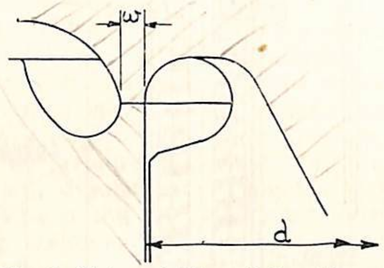

Fig. 4—Enlarged View of Seat Face.

The flat seat, although possessing certain disadvantages, is widely adopted on spring loaded valves principally in order to obtain the maximum discharge area for a given lift. There are no strict rules governing the actual width of the seating surface (w) but experiments have shown that for lapped metal to metal surfaces a contact pressure of at least 1·25 times the internal static pressure is required to ensure steam tightness. Consequently, if the valve is to reseat tight at the point of blowdown, then the seating width must be limited to a definite maximum value as follows :—

Let p = safety valve blow-off pressure.

Then load exerted by spring $= p \times \pi d^2/4$.

Assuming 4% blowdown, the steam pressure when valve reseats $= \cdot 96p$.

$\therefore$ The steam load at this point $= \cdot 96p \times \pi d^2/4$.

$\therefore$ Excess load available for reseating valve

$$= \text{spring load} - \text{steam load}.$$
$$= p \times \pi d^2/4 - \cdot 96p \times \pi d^2/4$$
$$= \cdot 04p \times \pi d^2/4.$$

Since seating width (w) is very small, seating area $\simeq \pi\, dw$.

$$\therefore \quad \text{Contact pressure} = \frac{\cdot 04p \times \pi d^2/4}{\pi\, dw}$$
$$= \cdot 01\ pd/w.$$

But this contact pressure must equal 1·25 times steam pressure.

$$\therefore \quad \cdot 01\ pd/w = 1{\cdot}25p$$
$$\therefore \quad w = \cdot 008d$$

The equivalent value for a 45° bevel seat $= \cdot 0088d$.

Theoretically the smaller is w the better but minimum values are, of course, limited by the compressive strength of the valve and seat materials.

Valve Lift.

Apart from mechanical reliability the capacity of a safety valve is the most important indication of its efficiency as a safety device. The capacity of a safety valve discharging steam at any given inlet pressure and temperature conditions is dependent principally on the discharge area available, which in turn is governed by two factors—the valve diameter and the distance the valve lifts from its seat. In order to obtain maximum discharge area the valve must lift to a height equal to $\frac{1}{4}$ of the valve diameter in the case of a flat type seat and approximately $\frac{1}{3}$ of valve diameter for a 45° seat.

Now it might be expected that once the valve lifts at the blow-off pressure the steam, acting on the additional exposed valve area represented by the seat width w, would force the valve wide open to give full bore discharge area. This, unfortunately, is not the case and in actual fact the lift is very often extremely limited, rarely exceeding 1/24 of the seat bore diameter on ordinary type spring loaded safety valves. This apparent resistance to full valve lift may be attributed chiefly to the action of the following factors :—

1. **Drop in steam load.**—As the valve discharges, a proportion of the potential energy represented by the pressure of the static steam changes into kinetic energy in the immediate vicinity of the annular orifice. The accompanying decrease in pressure reduces very slightly the total effective lifting load on the valve.

2. **Back pressure.**—Any back pressure which may be present in the valve chest will, by its action on the top of the valve, tend to create a downward load opposing valve lift.

3. **Variation in spring load.**—On spring loaded valves an inherent disadvantage arises from the slight compression of the spring which occurs during valve lift. Since the load exerted by a spring is proportional to its compression any additional compression will cause a corresponding increase in spring load. Thus, as the valve lifts to its maximum height, the spring load opposing the upward steam pressure increases proportionately. This is further aggravated by the fact that stiff springs are desirable in order to eliminate feathering during blowdown. (Numerous designs intended to compensate this variation in spring load have been experimented with since the early days of

safety valve history, but since they all basically rely on the incorporation of some form of lever or toggle device the added friction has been found to more than outweigh the advantage of constant spring load).

Dynamic effects of the escaping steam may also influence valve lift to some extent.

As indicated above, discharge capacity is directly dependent on valve lift, a safety valve possessing small lift consequently has a corresponding low discharge efficiency. The question of valve lift becomes particularly important on boilers having high evaporative capacity where considerable economy may be effected by the employment of large capacity safety valves, and the desire to attain improved lift has had a marked bearing on safety valve development.

Reaction Assisted Lift.

Perhaps one of the most universally adopted devices for improved valve lift is the incorporation of a lip around the periphery of the valve, a typical example of such a valve being shown in Fig. 5. It was realised early on that the reaction of the escaping steam against the increased valve area offered by a projecting lip could be used to assist the normal steam load acting on the main valve face. One of the earliest safety valves having this improvement

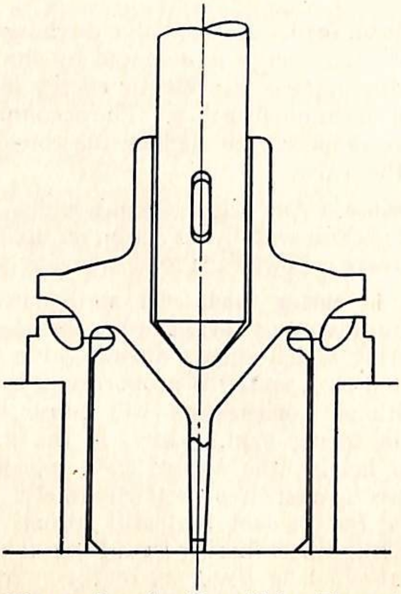

Fig. 5—Reaction Type Valve Element.

was introduced in 1848 by an Englishman named Charles Ritchie, whose covering patent specification included the following description :—

> "As soon as the pressure of the steam raises the valve from its seat the lip, being exposed to the pressure of the steam presents an increased surface which compensates for the increasing resistance of the helical spring."

A similar device was incorporated in the Naylor valve, a later improved lift type safety valve, patented by William Naylor in 1863. Fig. 6 shows a section of this valve. It will be seen that the valve lip curves downwards below the level of the seat, thus causing the escaping steam flow to turn through nearly 180°. A second feature peculiar to the Naylor valve was the compensation of the increasing springload during lift by means of the varying leverage offered by a bent lever connecting the spring with the valve spindle.

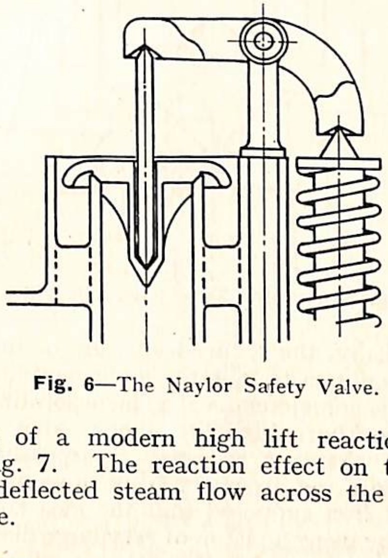

Fig. 6—The Naylor Safety Valve.

An example of a modern high lift reaction type valve is illustrated in Fig. 7. The reaction effect on the valve head is caused by the deflected steam flow across the underside of the adjustable sleeve.

Piston Assisted Lift.

When a safety valve blows the steam pressure does not fall to atmospheric immediately upon leaving the valve discharge orifice and in practice the pressure in the valve exhaust chest is usually found to be something like 1/10 of the blow-off pressure. Following safety valve trials on British destroyers in 1908, it was realised that this back pressure could be usefully employed to assist the valve to lift during discharge by allowing it to act on a secondary piston located immediately above the valve. Fig. 8 shows the

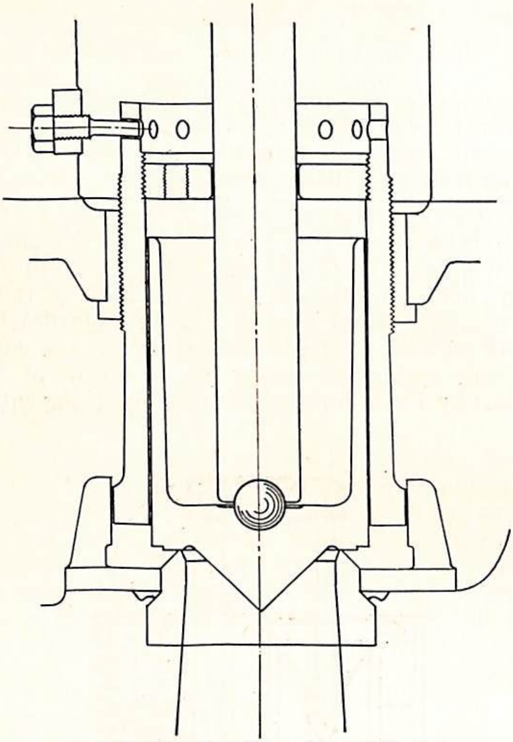

Fig. 7—Reaction Type High Lift Valve.

idea diagrammatically, the reduced pressure of the exhaust steam acting on the piston area to give the additional lifting load, whilst Fig. 9 illustrates a commercial valve incorporating this feature—the well known Cockburn high lift safety valve. In such safety valves the outlet chamber is usually purposely restricted just sufficiently to ensure the necessary back pressure.

It might be at first supposed that the idea could be exploited to full advantage by using a piston of very large diameter in relation to valve size in order to obtain the maximum lift possible. Such an arrangement would certainly secure the desired lift but the question of closing the valve is then introduced, since the additional load used so effectively to assist lifting acts as a resistance to the spring load during blowdown and if a large piston is employed the valve will not reseat tight until the boiler pressure has dropped considerably below the valve setting. The resulting steam loss alone nullifies any advantage gained by increased valve capacity.

The blowdown pressure range, *i.e.*, the range between blow-off pressure and the pressure at which valve recloses, is limited by

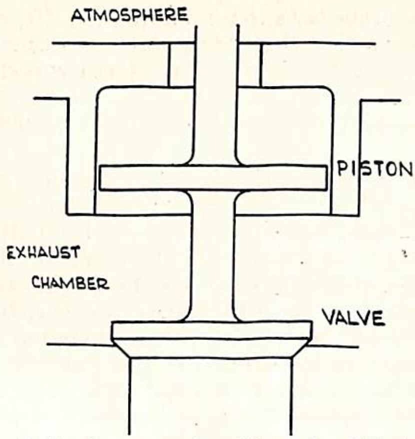

Fig. 8—Valve Incorporating Piston for Lift Assistance.

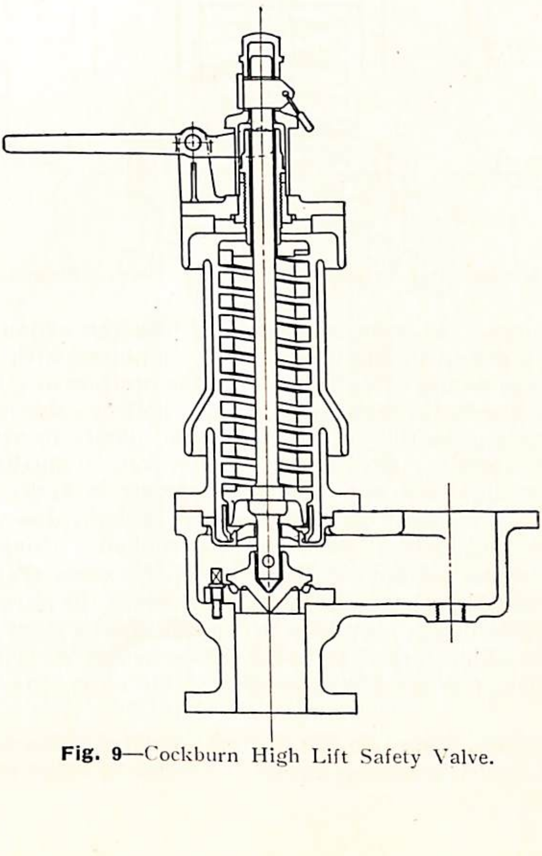

Fig. 9—Cockburn High Lift Safety Valve.

B.S. 759 and many other laws to a maximum of 5% of the relieving pressure; this means that if a safety valve is designed to blow at 500 lbs./sq. in. for example, the valve should reseat at a pressure not lower than 475 lbs./sq. in. At high pressures the value of 5% would be uneconomical and 3% is a more generally accepted figure.

As has already been stated, on piston assisted high lift valves the blowdown may present some difficulties due to the additional lifting load which comes into play as soon as the valve lifts from its seat, but just as the back pressure in the exhaust chamber may be utilised to provide extra lift, so it can be employed to assist the valve to reseat tight. Fig. 10 illustrates the basic operating principle of an assisted blowdown device commonly incorporated in high lift valves. At position (*a*) the valve is shown before

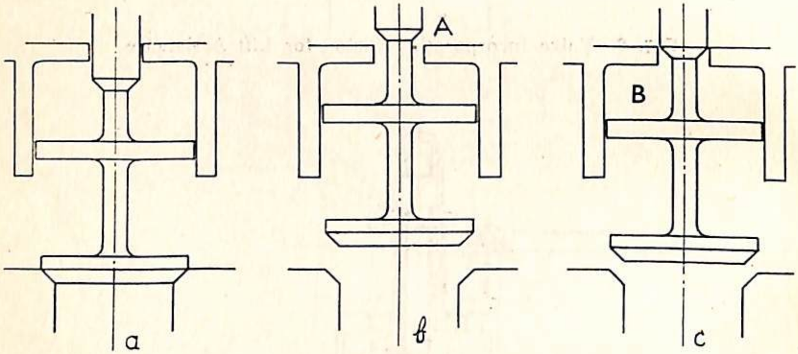

Fig. 10—Operating Principle of Pressure Assisted Blowdown.

discharge occurs; as soon as the valve lifts the exhaust steam, acting on the piston in the normal way, combines with the main steam load to blow the valve full open to the position at (*b*). Meanwhile, steam bleeds through the clearance between the piston and guide and, so long as the valve is full open, passes to atmosphere through the annular opening 'A.' However, immediately the valve drops a slight amount (due to a decrease in boiler pressure) to position (*c*), the collar on the spindle is brought down into the opening at 'A' and further steam escape is choked. Consequently, the exhaust steam occupying chamber 'B' becomes trapped and exerts a downward pressure on the piston to assist the normal spring load during blowdown. It will be readily appreciated that the position of the collar in relation to the escape orifice 'A' is important and in practice it is usually made adjustable to facilitate correct presetting.

The positive action resulting from assisted blowdown also reduces erosion of the seating faces. The closing takes place from

nearly full lift of the valve and the steam discharge is cut off sharply, thus protecting the seat from the wire drawing action which accompanies a low settling close.

Pop Safety Valves.

In America considerable attention has been directed to the development of the high lift pop safety valve ; the word "pop" is a self-explanatory description of the explosive type discharge characteristic of these valves.

The first successful spring loaded safety valve having this popping action was invented in 1866 by an American, George W. Richardson. The principle behind his design (shown in Fig. 11) is used to the present day and is the foundation on which an important class of modern high duty safety valves is constructed. In the original Richardson valve, as will be seen from the acconpanying illustration, the valve was designed with an adjustable peripheral lip subtending below the seat to form a secondary chamber

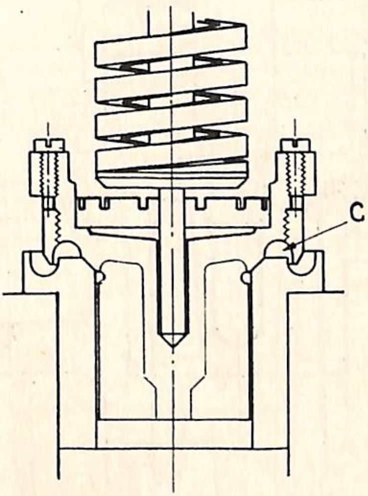

Fig. 11—The Richardson Valve.

(*c*). As the valve lifts, the discharged steam enters this annular chamber but escape is restricted by the projecting lip. The resulting intermediate pressure acting on the lip area creates an unstable condition and forces the valve to lift high to a position of equilibrium. The action takes place almost instantaneously to give the explosive type discharge.

In present day designs the adjustable ring is positioned on the seat element and is locked after setting by means of a set screw

suitably fixed to the main valve chest as in Fig. 12. The annular chamber is located between the ring and valve lip at (C) and the inner section of the ring is of conical form. Immediately the valve lifts discharged steam enters 'C' but further passage is partially restricted at 'A' and the intermediate pressure in 'C' causes the valve to lift higher. This increase in lift allows a greater quantity of steam to pass, but the diameter of the restricting orifice increases also (due to the diverging conical section) to balance the greater flow. Thus, on discharge, the valve is rapidly blown to a position of equilibrium where the steam pressure distribution is just sufficient to balance the spring load.

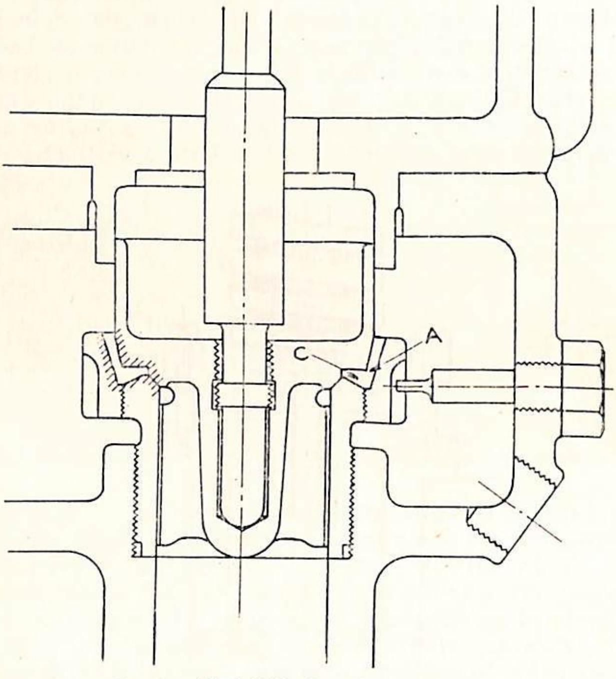

Fig. 12—High Lift Pop Safety Valve.

The behaviour of steam expanding from a discharge orifice is extremely complex and in order to obtain optimum conditions for full lift and effective blowdown the profile of the discharge passage (shaded surfaces in Fig. 12) must conform to precise dimensions. Also, since the ring governs the degree of restriction at 'A' its position relative to the valve is critical. For it will be readily appreciated that if the ring is too low there will be insufficient restriction in the secondary chamber and maximum lift will be

unobtainable, whilst on the other hand, if the ring is too high the lifting force will be excessive and blowdown will become a difficulty.

Fig. 13 illustrates the principal operating features of a more recent valve designed for improved capacity. From a study of the illustration it will be seen that this valve incorporates the Richardson adjusting ring and the collar type blowdown adjustment whilst the upper part of the valve serves for guiding purposes as well as piston for assisted lift and blowdown. The top guided form of valve has certain advantages over the winged type especially on safety valves possessing full lift where guides in the seat bore would interfere with discharge area. In the design shown, small bleed holes located at the base of the valve allow steam to pass through to the blowdown chamber. Where the Richardson ring is combined with the collar type blowdown regulation, as in the above valve, correct setting for full lift and restricted blowdown may call for skilful adjustment. Tests have shown that alteration

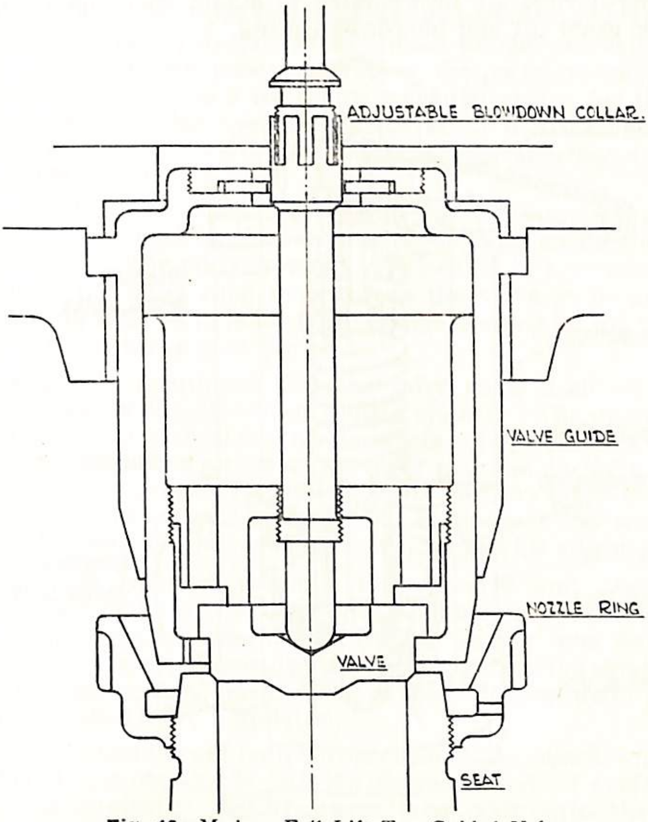

Fig. 13—Modern Full Lift Top Guided Valve.

of either the blowdown collar or the adjusting ring by as little as 1/10 of a turn (giving less than ·01″ vertical movement) may affect considerably the lift and blowdown characteristics of that particular valve.

Micrometer Adjustments.

The desire to market a direct spring load safety valve capable of discharging the greatest possible quantity of steam through a limited seat diameter under given pressure and temperature entrance conditions, together with the minimum blowdown range, has resulted in the introduction of two important designs within recent years—the Foster Type 38-SV, and Dewrance-Consolidated Maxiflow safety valves. These valves, although differing in details possess a certain fundamental similarity in operating principle and a brief description of both types will be given here. Each valve utilises the conventional nozzle ring but additional regulating devices are incorporated to obtain micrometer adjustment for exact lift and blowdown control.

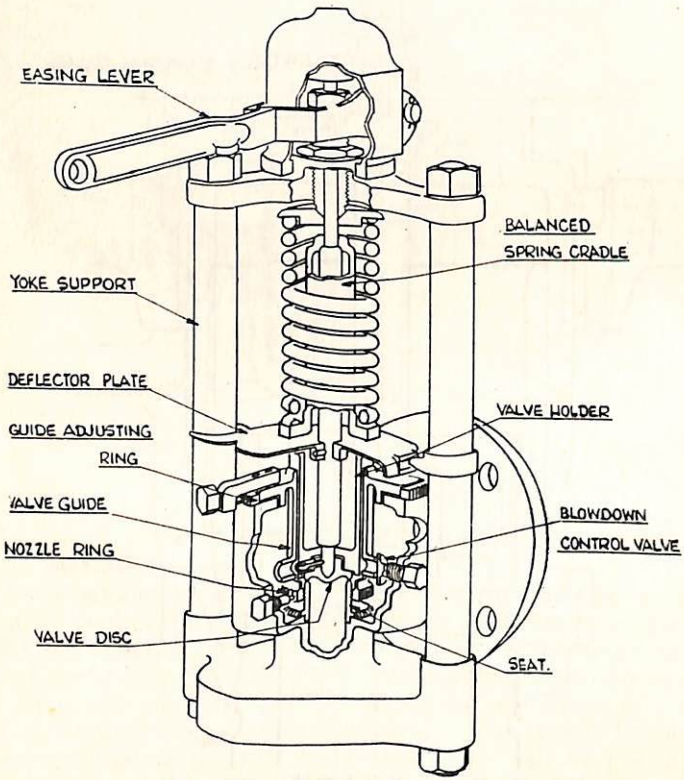

Fig. 14—Foster Type 38 Safety Valve.

Fig. 14 shows a cut-away view of the Foster valve. Unorthodox in appearance it possesses several features which deserve mention. To facilitate self-alignment the actual valve disc is a separate element fitting loosely in the main disc holder; the disc holder also provides the lip area for steam reaction effect and serves for guiding purposes. The valve guide projects down to the base of the disc holder and, in conjunction with the nozzle ring, forms the secondary restricting orifice for assisted valve lift. Around the skirting at the base of the guide there are a number of small ports which are located in line with the exhaust steam belt and serve as a steam byepass during discharge. The guide may be adjusted vertically by means of a screwed ring above the valve chest, whilst the outlet area of each port may be individually regulated by valves suitably positioned around the valve chest. In setting the guide is so positioned that during the initial stages of valve lift the byepass ports are covered by the disc holder; upon valve discharge they are thus momentarily inoperative and the full reaction effect of the steam may be utilised to blow the valve wide open. In the full lift position, however, the ports are completely uncovered and provide a secondary regulated escape for the discharged steam. This byepass has the effect of reducing steam reaction load on the valve and, consequently, assisting a sharp, tight valve closure.

In the Maxiflow valve, illustrated in Fig. 15, blowdown control is similarly obtained by the regulated byepass of steam from the exhaust belt. The ports, however, are located in a separate ring (known as the outer ring) mounted on the valve guide and are positioned in relation to the valve discharge opening by the vertical adjustment of this ring on the guide.

It might be mentioned that the valve guide itself serves no other purpose in the blowdown control system. The discharging area of each port is regulated by the single adjustment of a second ring (trim ring) which screws on the outer ring and partially covers the exit of each port. The third control element takes the form of the conventional nozzle ring located on the seat and which, in conjunction with the outer ring, provides for full lift adjustment.

A feature common to both Foster valve and Dewrance-Consolidated valve is the twin strut method of yoke support. This design provides considerable weight saving over the conventional flanged type cover and moreover assists in the elimination of what is termed the "crawl" effect of blow-off pressure (a phenomenon described more fully later).

The manufacturers of both valves claim that a blowdown range as low as 1% is obtainable with the above methods of micrometer control adjustment; this compares favourably with the more usual 3%.

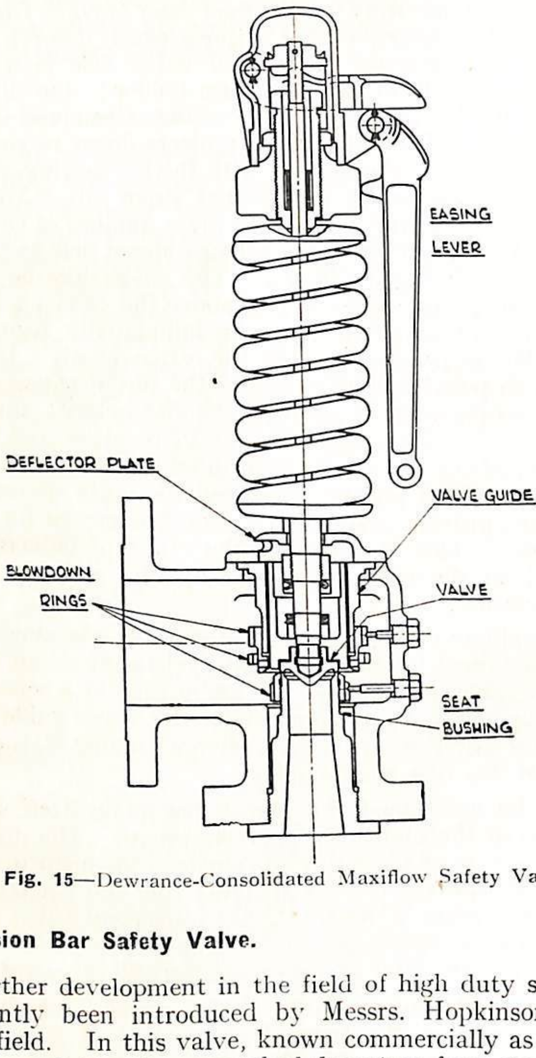

Fig. 15—Dewrance-Consolidated Maxiflow Safety Valve.

The Torsion Bar Safety Valve.

A further development in the field of high duty safety valves has recently been introduced by Messrs. Hopkinsons, Ltd., of Huddersfield. In this valve, known commercially as the "Hylif" torsion bar safety valve, a marked departure from standard safety valve design is shown by the adoption of a torsion bar for valve loading in place of the normal helical spring.

This torsion bar, which is of uniform circular section, is mounted in a horizontal position immediately above the valve chest. To obtain a balanced action the safety valve is constructed with twin torsion bars both of which load a single central valve. Load adjustment is achieved by means of an adjusting screw operating

through toggle arms connected to one end of each bar, whilst lever arms connected to the opposite end transmit the load to the valve head.

The inherent advantage possessed by torsion bar loading lies in the ability to control within accurate limits the final dimensions of the loading rod and thus ensure uniform exertion of the design load on the valve at all stages of valve opening. As is well-known, a helical spring must be completely formed before it can be heat treated to give the desired temper, this means that any deformation in the spring arising from the heat treatment will remain since no further dimensional alterations can be made to the spring once this operation is completed. Due to the shaping of the ends a helical spring may also exert uneven loading on the valve head, thus causing a slight canting over when the valve opens. Torsion bars, on the other hand, can be obtained of uniform composition and ground accurately to size as a final operation after heat treatment. Thus full control over final load characteristics can be assured.

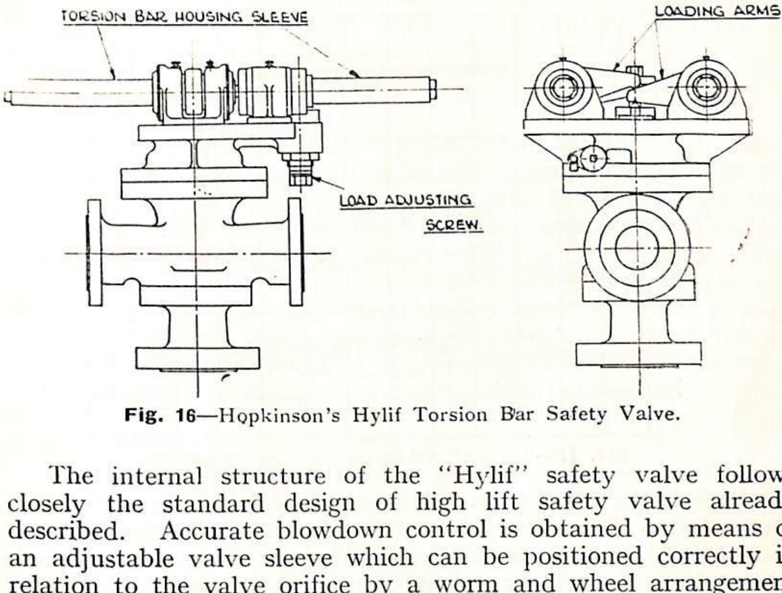

Fig. 16—Hopkinson's Hylif Torsion Bar Safety Valve.

The internal structure of the "Hylif" safety valve follows closely the standard design of high lift safety valve already described. Accurate blowdown control is obtained by means of an adjustable valve sleeve which can be positioned correctly in relation to the valve orifice by a worm and wheel arrangement with micrometer adjustment. Twin outlets are provided for balanced steam discharge.

Torsion bar safety valves are designed primarily for operating pressures above 900 lb./sq. in. and they are now being increasingly adopted on boilers of high evaporative capacity as typified in modern power station installations.

Crawl.

Safety valves discharging high temperature steam are sometimes affected by slight variations in valve blow-off pressure, the effect (known as crawl) becoming most marked on valves blowing at short consecutive intervals. When high temperature discharged steam comes into contact with the components (*e.g.*, body, cover, spring, etc.) located on the outlet side of the valve their physical dimensions are slightly altered by the resultant expansion which occurs. In addition, any heat reaching the spring will cause a drop in its modulus of rigidity with a consequent slight reduction in spring strength. The overall effect of the temperature rise is to decrease the effective loading on the valve head and so lower the blow-off pressure. If the boiler pressure is sufficiently reduced by the first blow-off then no further discharge will occur and the safety valve will cool down to its normal temperature, but when pressure remains high the valve will continue to pop at pressures well below the initial setting value. The graph shown in Fig. 17 illustrates the crawl effect on a safety valve discharging steam at

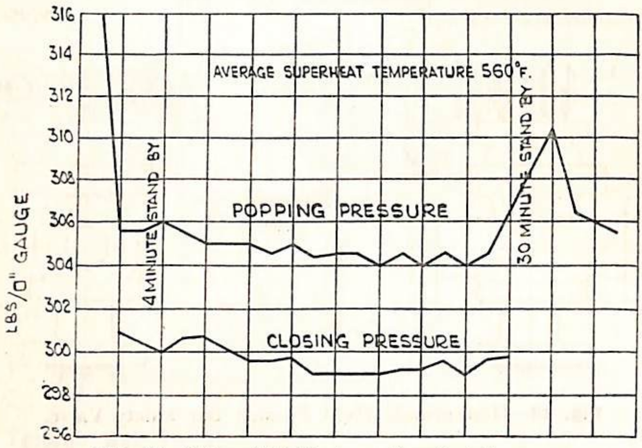

Fig. 17—"Crawl" Effect on Popping Pressure.

approximately 316 lbs./sq. in. superheated to 560°F. It will be noticed that after a 30 minute standby, in which the valve has time to cool down from the previous blow-off, the popping pressure shows definite signs of returning to the original set pressure.

A logical remedy that is widely adopted for high temperature work is the employment of materials which, while still suitable for their primary task, remain to a large extent unaffected by temperature rises occurring during valve discharge.

In this direction the introduction of Invar steel has been found to give good results. Invar is a high nickel steel possessing

dimensional stability over a wide range of ambient temperatures, expansion of the metal is, in fact, practically nil up to 250°F. Trouble arising from loss of spring rigidity at elevated temperatures is effectively reduced by the selection of suitable steels, notably tungsten alloys, for their manufacture. Also, by using alloy steels possessing a high coefficient of expansion for the valve spindle, reduction in spring strength may be compensated by the slight spring compression resulting from expansion of the spindle during discharge. An additional feature incorporated in certain safety valves is the provision of a plate immediately beneath the lower spring support to deflect any escaping steam downwards and away from the spring.

It might be argued, on the basis that prevention is better than cure, that the simplest method of crawl elimination would be to prevent entirely the escape of steam through the valve chest. Such an idea, however, would necessitate the incorporation of a gland at the point where the main valve spindle passes through the valve chest, a practice which should be avoided at all costs since the restrictive effect on freedom of spindle movement due to friction at the gland entirely outweighs advantages achieved by steam tightness.

The "Thermodisc."

A troublesome effect often experienced on spring loaded valves is the tendency for the valve to breath or leak slightly when the steam pressure is approaching the popping pressure ; the wire drawing action of the escaping steam may cause excessive erosion of the seating surfaces and shorten their life considerably. This leakage, which may take place even with perfectly ground-in seat faces, occurs when the differential loading on the valve (*i.e.*, excess of spring load over steam load) is insufficient to overcome minute distortion resulting from slight temperature variations at the valve or the presence of microscopic water solids between the contact surfaces, and it almost invariably disappears when the steam pressure drops.

Recent investigations concerning the mechanics of valve leakage have indicated that a contributory cause arises from the thermal effects occurring at the seating face. The steam flow resulting from any initial leakage, being throttled by the minute escape passage, expands as an imperfect gas and a sharp temperature drop occurs at the point of leakage. This temperature drop causes local refrigeration of the seating surfaces and increased distortion is produced by the consequent temperature gradient formed around the circumference.

Fig. 18 shows the results of tests made on an actual valve under operating conditions. For the purpose of the test a number of

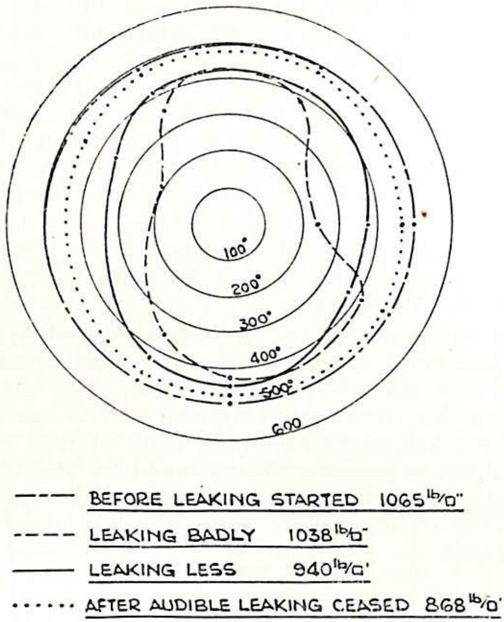

Fig. 18—Temperature Gradient around Seat Face during Valve Leakage.

thermocouple beads were located round the seating face of both the valve and seat elements, temperature readings being taken at various stages during valve leakage. It will be noticed that before leaking commenced the temperature around the contact face was fairly uniform at about 500°F., whilst at a pressure of 1038 lbs./sq. in. when the valve was leaking badly a temperature gradient as high as 250°F. existed around the seating face. The results of such tests indicated the necessity to provide some means of equalising these temperature variations in order to combat valve leakage. The problem was accentuated by the poor heat conductivity possessed by the stainless steels normally employed for valves and seats on high temperature work. A design was eventually evolved in which the valve seat face was undercut to allow easier heat conduction from the steam to the seat face through the reduced metal section, thus providing a more uniform temperature distribution over the face. This design, in its commercial form, the Dewrance-Consolidated "Thermodisc," is shown in Fig. 19. An objection to this type of valve may arise from the apparent weakness of the thin lip, but a liberal safety factor is provided and no troubles arising from fracture or distortion have been experienced in practice. In Fig. 19, the undercut is located in the valve disc ; it could alternatively be machined in the seat

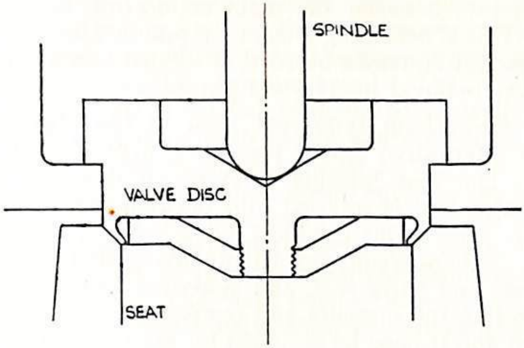

Fig. 19—Dewrance-Consolidated Thermodisc.

element, or again, in the case of a flat seat, the valve could be undercut externally, in which case the air on the outlet side of the valve would act as the temperature equalising medium.

Grooved Valve Guide.

Experience with safety valves operating at high pressure has shown that a tendency for the valve element to rotate occurs during normal discharge. This rotational effect, which may cause excessive wear at the ball seat point on the main spindle and promote galling between the valve and guide, has been traced to the escape of small quantities of steam through the clearance between valve and guide. Tests conducted in the U.S. have indicated that this steam, in escaping, follows a spiral path around the valve and the resulting unequal distribution of pressure between the valve and guide gives rise to the rotational action of the valve.

To counteract the phenomenon it has been necessary to devise some method whereby the unequal pressure distribution occurring in the guide clearance is eliminated and it is specifically for this purpose that the grooved guide has been introduced. A series of concentric grooves machined in the valve guide allow pressure equalisation to take place around the valve for the whole length of the guided surface and it has been found in practice that by this method the tendency for valve rotation may be effectively neutralised. The uninterrupted steam contact around the guide provided by this design also allows full temperature equalisation and thereby affords uniform expansion of the valve and guide elements.

Seat Design.

Referring again to Figs. 14 and 15, certain aspects of the seat designs shown deserve mention. In the Foster valve the seat is a push fit in the body and is welded in position, thus eliminating

all pressure joints between the main connecting flange and the valve seat. This is becoming a common practice for high pressure work but it should be remembered that welded joints considerably complicate seat removal for renewal purposes.

The through-bushing type seat used in the Maxiflow valve illustrates an alternative design possessing a number of inherent advantages. The seat bushing is extended down to the main connecting flange and is screwed into position from beneath, the shoulder at the base forms the joint facing with the mating flange on the boiler. There is consequently no break whatsoever between the main flange and valve seat, and moreover no part of the main body is subject to the pressure and temperature conditions of the internal steam and it need be designed for exhaust conditions only. The internal machined surface of the seat may be shaped to give streamline inlet flow conditions, an important aspect governing maximum capacity on safety valves where valve lift is sufficiently high to give full bore discharge area.

The principal drawback of the through-bushing design is the relatively high cost involved in the use of expensive materials, normally required for the seat surface only, for so large a seat element although this disadvantage is to some extent directly compensated by the more homogeneous and sound structure provided by the bushing at this highly stressed part of the safety valve.

Twin Exhaust Outlets.

On certain types of heavy duty safety valve, the exhaust chamber is provided with two outlet passages of equal area positioned diametrically opposite each other in the valve chest. When a large capacity valve discharges steam at high pressure considerable vibration is set up at the valve due to the reaction effect of the escaping steam which may be likened to the recoil action of a gun. By the provision of twin outlets the discharge reaction at the valve may be balanced and excessive vibration stress in the neck of the valve body eliminated.

A modified single outlet design intended to serve the same purpose is constructed with an internal baffle wall positioned at the entrance to the exhaust passage. This has the effect of damping the reaction force, set up by the sudden steam discharge, by the creation of a balancing load on the baffle.

Relay Safety Valves.

The final class of safety valve to be considered is the relay actuated type in which the pressure sensitive unit, instead of providing direct valve loading, serves merely as a pilot for the operation of the main valve.

The Cockburn "Fullbore" safety valve, used extensively on naval boilers, is perhaps one of the best known representatives of the relay type valve and embodies the general principles on which most other valves in this class are based. The "Fullbore" valve, shown in Fig. 20, has twin valves mounted centrally in the main body, a single inlet feeds both valves and discharged steam is led to atmosphere via the common exhaust at the top. Small ports located in the inlet passage allow steam entry to the control valves located either side of the main valve chest. These control valves are small bore safety valves of the orthodox spring loaded design which function solely as pressure sensitive elements for actuation of the main valves. The main valves themselves are located in a horizontal position in the valve chest and are loaded by the direct

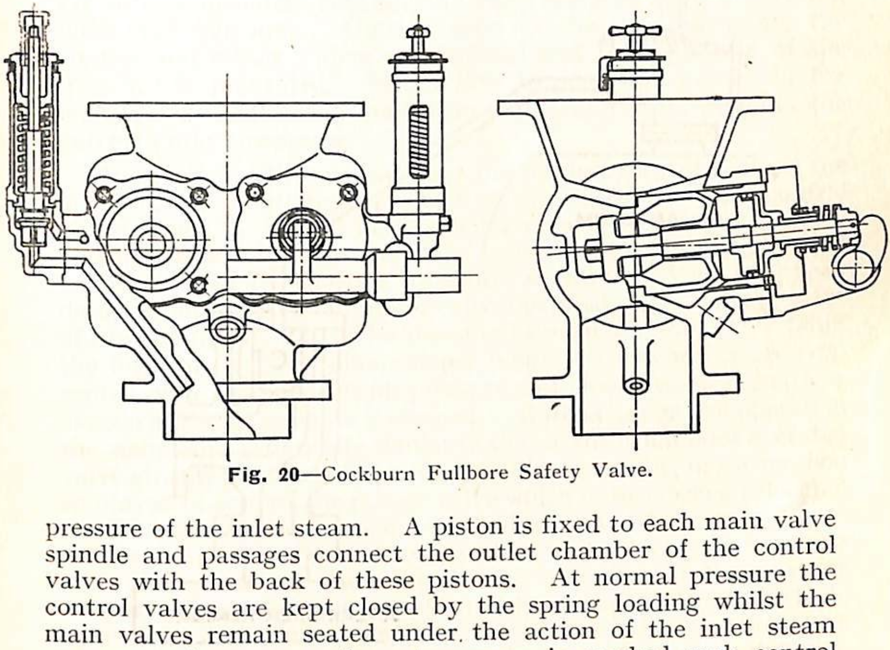

Fig. 20—Cockburn Fullbore Safety Valve.

pressure of the inlet steam. A piston is fixed to each main valve spindle and passages connect the outlet chamber of the control valves with the back of these pistons. At normal pressure the control valves are kept closed by the spring loading whilst the main valves remain seated under the action of the inlet steam pressure. As soon as the set pressure is reached each control valve blows and, in lifting, carries a plate against the underside of the valve casing. Escape to atmosphere is thus prevented and the discharged steam flows to the back of the main valve operating pistons. Since the piston diameter is greater than the main valve diameter an unbalanced load is set up and the valve is blown full open to give maximum discharge area, the escaping steam passing through ports in the spindle guide to the outlet passage. Valve reclosure functions in the exactly reverse manner, a drop in boiler pressure causing the control valve to close and

thereby release the small plate from its seating on the valve casing. The resulting steam escape from the outlet chamber of the control valve relieves the steam load behind the piston and main valve reclosure is effected by the combined action of the return spring and inlet steam pressure.

A more recent introduction in the field of relay safety valves for marine work is the Dewrance-Consolidated drum pilot operated valve, a diagrammatic view of which is shown in Fig. 21. In broad principles, this valve is similar to the "Fullbore" valve with the essential difference that the main valve functions as a super-

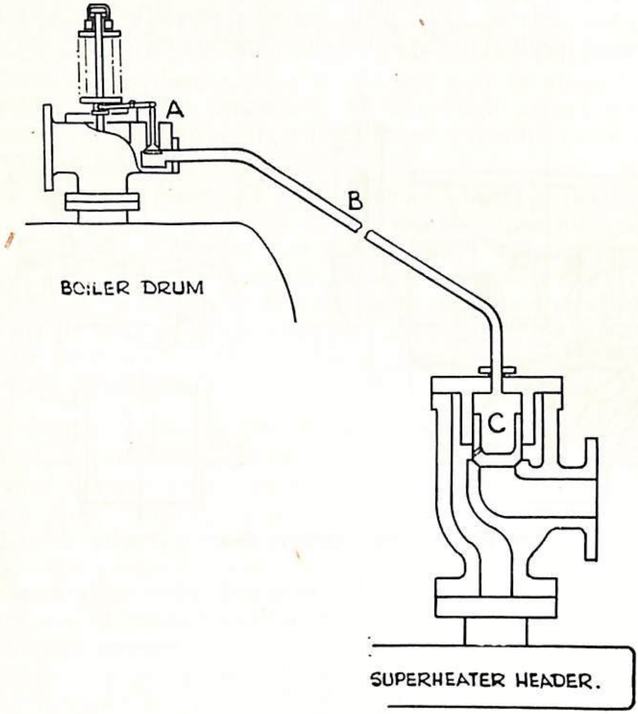

Fig. 21—Dewrance-Consolidated Drum Pilot Valve.

heater unloading valve whilst the control valve or pilot valve is an independent unit located on the boiler drum and serving the double purpose of normal drum safety valve and control actuator for the main valve. The main valve disc is a loose element held on its seat by the inlet steam pressure in chamber (C) and fits closely in the surrounding guide. Small bleed holes in the underside of the valve disc provide the necessary steam connection between the pressure chamber (C) and the inlet side of the valve. At the control

valve a small secondary release valve (A) is housed adjacent to the valve chest and is connected by means of a lever mechanism with the control valve spindle, such that any lift of the control valve will automatically cause the release valve to open. A steam pipe (B) connects between the release valve and the main valve pressure chamber (C).

In operation, the control valve functions as a normal drum safety valve and so long as it is closed the release valve (A) remains seated under the action of the steam pressure in (B). When the popping pressure is reached, however, the control valve blows and, in lifting, opens the release valve and thus provides an escape for the steam in chamber (C) via the connecting pipe. The steam load above the main valve is relieved and, under the action of the inlet pressure immediately beneath, the valve is forced open to give full bore discharge area. Upon a drop in the drum pressure, the control and release valves are reclosed and further steam escape from (C) is prevented. Steam flow through bleed holes in the main valve immediately builds up sufficient pressure behind the valve to effect reclosure.

It will be readily evident that the performance of each of the above valves is entirely dependent on the sensitivity of the control valves which themselves may be liable to such troubles as "crawl" and prolonged blowdown commonly associated with spring loaded valves. A more effective pressure sensitive device has been designed for use on an electrically operated relay safety valve illustrated in Fig. 22. The pressure element for this valve takes the form of twin beryllium copper bourdon tubes which are connected with the boiler drum pressure and close electrical contacts as soon as the set pressure is reached. Both in design and operation the main valve is basically similar to that of the drum pilot operated valve already described, the principal difference lying in the method employed to actuate the release valve which in this design is located immediately above the main valve disc. An electrical solenoid unit is attached to the main valve chest and a lever arm connects with the spindle of the release valve. Closure of the pressure element contacts (at the set pressure) energises the solenoid which, by means of the lever arm, lifts the release valve to provide an escape for the steam in the pressure chamber above the valve disc. The high degree of sensitivity obtainable with the bourdon tube pressure element enables the safety valve to be set within 2% or 3% of the working pressure whilst a blowdown as low as 1% is possible. It should be pointed out that the release valve can be operated only by the solenoid since the return spring above the valve is actually loaded several times above the highest steam load obtainable. The release valve consequently remains unaffected by the pressure in the main valve and, in the event of an electrical power failure, the whole safety valve becomes inoperative. The

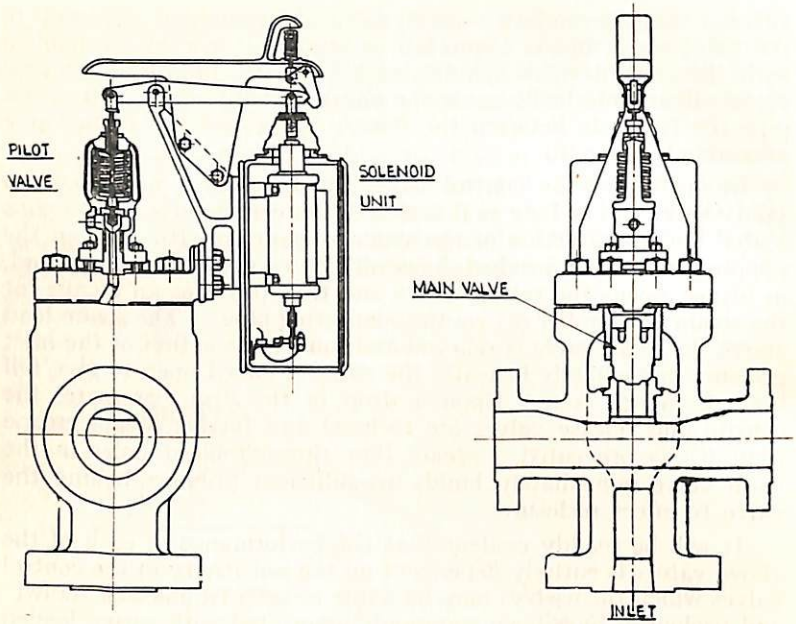

Fig. 22—Electrically Operated Power Control Valve.

possibility of a power failure prevents the inclusion, under British Boiler Laws, of electrically operated valves when assessing the total discharge area necessary for a boiler and such valves as the solenoid operated type just considered may be employed only as additional to the minimum requirements of the covering specification.

A modified design introduced specifically to overcome this disadvantage incorporates the relay unit of the drum pilot valve. The drum valve popping pressure is set slightly above the electrical pressure unit setting so that under normal conditions the solenoid functions as main valve actuator but in the event of power failure valve operation is effected by the drum pilot valve.

The relay safety valve possesses definite inherent advantages over its spring loaded counterpart and considerable economy may be attained by its employment on high duty boilers. The dispensation of bulky springs and accompanying heavy top gear associated with spring loaded valves provides a marked saving in weight whilst the pressure method of valve loading universally adopted eliminates breathing and ensures steam tightness right up to the blow-off pressure. Furthermore, the discharge capacity of these valves is very high and a single valve is usually sufficient to cope with all minor pressure variations in the boiler ; consequently,

by a suitable arrangement of valve settings the remainder of the boiler safety valves need rarely be called upon to function, thereby reducing wear to a minimum.

Valve Capacity.

The importance of discharging capacity as a measure of safety valve efficiency has already been stressed elsewhere in this pamphlet. In order to ensure the provision of adequate relief to full boiler load the discharge rating of every safety valve must be known and, since individual testing is out of the question, a reliable standard formula based on known valve dimensions is of prime necessity.

The flow of steam at constant inlet pressure p, through any given orifice is dependent on the pressure differential across the orifice and increases in quantity with decrease in outlet pressure p_2 until such time that p_2 has dropped to a value equal to $\cdot 57p$, after which no further increase in steam flow will result from a continued drop in p_2 and the flow rate is said to be "critical." Steam flow in the critical region obeys the adiabatic equation :—

$$M = \cdot 3003\ a \sqrt{\frac{p_1}{V_1}}$$

where M = flow rate of saturated steam in lb./sec.*
a = discharge area of orifice in sq. ins.
p_1 = inlet steam pressure lb./sq. in. abs.
V_1 = specific volume of inlet steam cu. ft./lb.

For the purpose of theoretical capacity assessment the annular discharge orifice of a safety valve may be regarded as a nozzle having a flow area '*a*' equal to the developed section at the point of greatest constriction (*i.e.*, '*x*' in Fig. 23) and the above equation may be applied to find the steam flow under given entrance con-

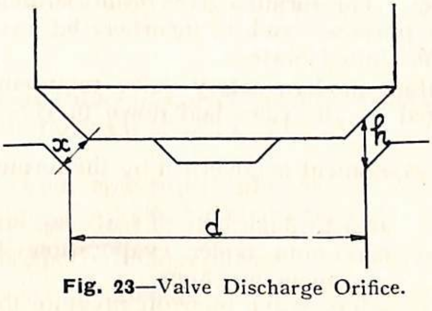

Fig. 23—Valve Discharge Orifice.

* For superheated steam $M = \cdot 3155\ a \sqrt{\frac{p_1}{V_1}}$

ditions. Due to the presence of friction, turbulence, etc., this formula must be modified for practical purposes to include a discharge coefficient k_d such that :—

$$M = \cdot 3003\ a\ k_d \sqrt{\frac{p_1}{V_1}}$$

Values of k_d usually lie between ·75 and ·9.

The orifice area 'a' $= \pi\, dh$ in the case of a flat type seat whilst for a 45° bevel seat 'a' $= 2{\cdot}22\, dh + 1{\cdot}11\, h^2$.

Consider now the case of a high lift safety valve having a 45° bevel seat of 3″ bore diameter and a known valve lift of ·33″. If this valve is set to discharge saturated steam at 600 lb./sq. in. gauge then its theoretical capacity may be determined from the above formula as follows :—

$$a = 2{\cdot}22 \times 3 \times {\cdot}33 + 1{\cdot}11 \times {\cdot}33^2 = 2{\cdot}32 \text{ sq. ins.}$$
$$p_1 = 615 \text{ lb./sq. in abs.}$$
$$V_1 = {\cdot}75 \text{ cu. ft./lb.}$$

$$\therefore\ M = {\cdot}3003 \times 2{\cdot}32 \times \sqrt{\frac{615}{{\cdot}75}}$$
$$= {\cdot}3003 \times 2{\cdot}32 \times 28{\cdot}6$$
$$= 20 \text{ lb./sec. or } 72{,}000 \text{ lb./hr.}$$

The actual tested capacity of this particular valve is known to be 63,000 lb./hr., the coefficient of discharge k_d may consequently be established from the ratio :—

$$\frac{\text{Actual capacity}}{\text{Theoretical capacity}} = \frac{63{,}000}{72{,}000} = {\cdot}875$$

A simplified derivation of the adiabatic flow equation, developed by Napier, gives $M = \frac{a\ p_1}{70}$ where M, a and p_1, have the same meaning as before. This formula gives results sufficiently accurate for all practical purposes and is incorporated in the A.S.M.E. Boiler Code of the United States.

In Great Britain modern safety valve requirements for land boilers are covered by the rules laid down in B.S. 759, 1950, in which valve size assessment is governed by the formula $A = \frac{E}{CP}$

where A = area through bore of seat—sq. ins.
E = maximum boiler evaporation—lbs. saturated steam per hour.
P = safety valve blow-off pressure lb./sq. in. abs.
C = a constant which is dependent on the class of safety valve, viz. :—

Type of Valve	Lift (D = diameter)	C Spring Loaded Valves	C Weight Loaded Valves
Ordinary lift valve	D/24	4	4·8
High lift valve	D/12	8	9·6
Full lift valve	D/4	16	16

For water tube boilers of evaporative capacity greater than 10,000 lb. of water per hour C may be increased as follows :—

Type of Valve	(D = diameter)	C Spring Loaded Valves	C Weight Loaded Valves
Ordinary lift valve	D/24	4·8	4·8
High lift valve	D/12	9·6	9·6
Full lift valve	D/4	20	20

A superheat correction factor k_s derived from the ratio of saturated to superheated steam flow equations

$$\left(i.e., \ \frac{\cdot 3003\, a \sqrt{p_1/V_1}}{\cdot 3155\, a \sqrt{p_1/V_{s1}}} \right)$$

is incorporated for use in the case of valves discharging superheated steam, the necessary superheated steam area A_s being equal to $A \times k_s$ where A is the area calculated for a similar quantity of saturated steam and $k_s = \sqrt{1 + \frac{1\cdot 5T \text{ (No. degrees supt.)}}{1000}}$

A graph giving values of k_s over the most generally used pressure and temperature ranges is shown in Fig. 24.

Example.—Drum and superheater safety valves are required for a boiler having the following characteristics :—

Maximum continuous rating	100,000 lb./hr.
Drum steam pressure	650 lb./sq. in. gauge.
Superheater steam pressure	600 lb./sq. in. gauge.
Superheater steam temperature	750°F.

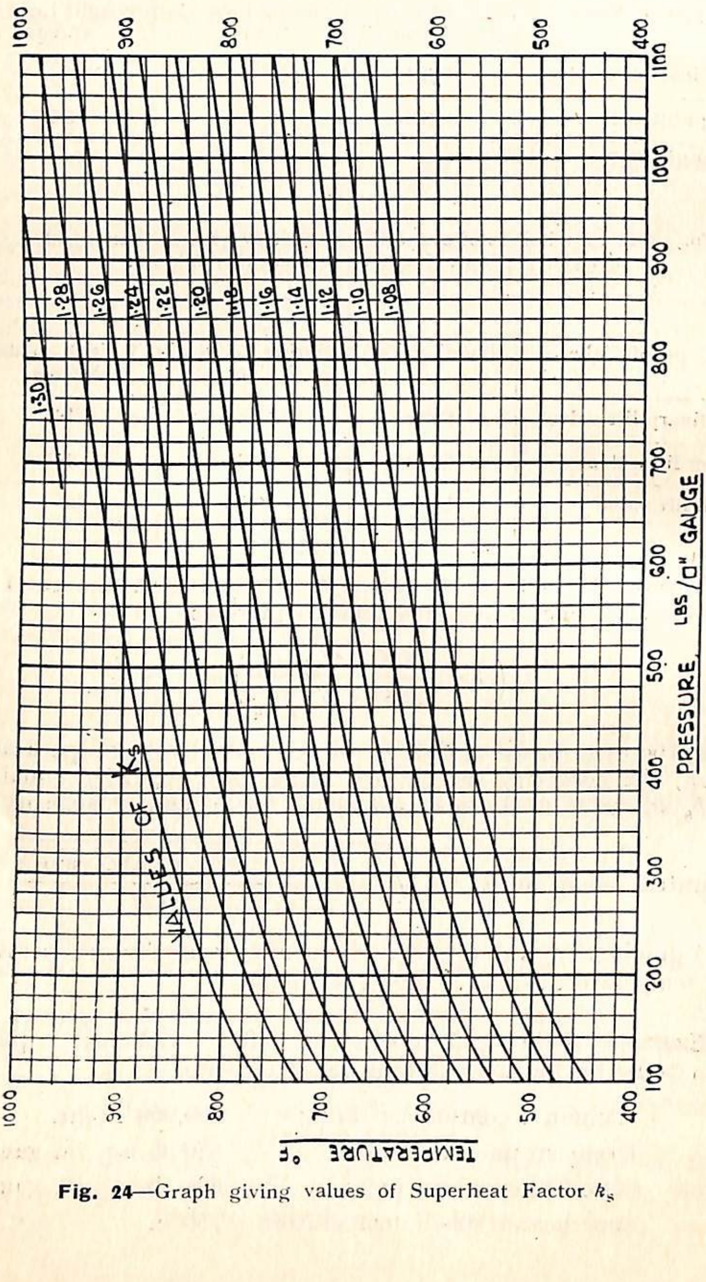

Fig. 24—Graph giving values of Superheat Factor k_s

Drum Valves.—Percentage of total evaporation passed by drum valves = 75%.*

$$\therefore\ E = 100{,}000 \times {\cdot}75 = 75{,}000 \text{ lb./hr.}$$

$$\therefore\ A = \frac{75{,}000}{9{\cdot}6 \times 665} = 11{\cdot}8 \text{ sq. ins.}$$

3 – $2\frac{1}{2}''$ bore high lift safety valves would be satisfactory.

Superheater Valves.—Percentage of total evaporation passed by superheater valves = 20%.*

$$\therefore\ E = 100{,}000 \times {\cdot}20 = 20{,}000 \text{ lb./hr.}$$

$$\therefore\ A_s = \frac{20{,}000}{9{\cdot}6 \times 615} \times 1{\cdot}18 \text{ (supt. constant obtained from graph).}$$

$$= 4 \text{ sq. ins.}$$

1 – $2\frac{1}{2}''$ bore high lift safety valve would be satisfactory.

Check for Total Capacity.—Total steam quantity discharge by combined area of drum valves = A.C.P. $= (3 \times 4{\cdot}9) \times 9{\cdot}6 \times 665$
$= 93{,}800$ lb./hr.

Total steam quantity discharged by combined area of superheater valves

$$= \frac{A_s\,CP}{k_s}$$

$$= \frac{4{\cdot}9 \times 9{\cdot}6 \times 615}{1{\cdot}18}$$

$$= 24{,}500 \text{ lb./hr.}$$

The total discharge capacity of the above safety valve arrangement is therefore equal to 118,300 lb./hr. and satisfactorily covers full boiler load.

The use of high lift valves in the above example provides an efficient arrangement for the boiler conditions given ; the choice of safety valve is not however tied down by any definite rules although the needs of economy and efficiency generally indicate the type of valve most suitable. On low pressure boilers of the Lancashire and Economic varieties, where evaporative capacities are generally very low, ordinary lift safety valves will provide quite adequate service. The extremely high evaporative capacities

* Division of the total evaporation into the ratios given is permissible under B.S. 759 provided total discharge area is sufficient to pass full boiler load.

met with on modern power station boilers, however, demand the employment of safety valves possessing correspondingly large discharge areas and for such purposes full bore spring or relay type valves become essential.

It is interesting to compare the valve capacity rating obtained from the B.S. formula with the figures derived from actual test measurements and the theoretical adiabatical flow equation. Consider for example the 3″ safety valve already dealt with and for which theoretical and actual flow values of 72,000 lb./hr. and 63,000 lb./hr. respectively were obtained. Under B.S. 759 this valve would be classed as a high lift safety valve and its assessed capacity at the given pressure of 600 lb./sq. in. could be calculated viz. :—

$$E = A.C.P.$$
$$= 7{\cdot}07 \times 9{\cdot}6 \times 615 = 41{,}700 \text{ lb./hr.}$$

A considerable discrepancy will be immediately apparent between the allowable rating under B.S. 759 and the valve's actual capabilities. This large safety factor provided by the British formula has attracted a certain amount of criticism from various sources since its inception but strong opinion in its favour has assured its retention and a number of countries (notably in the Commonwealth) now base their requirements on B.S. 759.

Entrance Conditions.

On ordinary lift and high lift safety valves where valve lift rarely exceeds about 1/10 of seat bore diameter, the orifice discharge coefficient k_d is governed almost entirely by the shape of the discharge passage immediately local to the valve opening and maximum steam flow is not appreciably affected by conditions at the safety valve entrance. On high capacity safety valves, however, where valve lift is sufficiently high to allow full bore discharge area, the throat diameter of the seat becomes the restricting orifice and the shape of the flow passage preceding the throat may noticeably influence the maximum discharge rate obtainable. A rounded entrance followed by a throat length of approximately 1 - 1½ diameters provide ideal flow nozzle conditions but design requirements inevitably necessitate a marked departure from this simple shape in practice. The inlet passage must, for example, pass through the thickness of the main connecting flange before reaching the valve opening; a relatively long throat is consequently inevitable and in order to reduce friction to a minimum a smooth, fully machined surface is essential. Furthermore, any sharp decrease in bore diameter, as might occur at the junction between safety valve and boiler connection, should be avoided by the provision of a suitable taper in order to maintain streamline flow.

Materials.

For safety valves operating under low pressure and temperature conditions the question of materials presents no serious manufacturing problems.

The use of cast-iron for safety valve bodies and covers is permissible for saturated steam pressures up to 150 lb./sq. in., whilst cast steel is universally employed above this pressure. At high superheat temperatures, however (850°F. and above), the problem of creep emerges and the introduction of special creep resistant alloy steels becomes essential, the $\frac{1}{2}\%$ Molyodendum alloys covered by B.S. 1398 provide examples of such steels.

The valve and seat components undoubtedly experience the most exacting conditions that exist in a safety valve during normal operation and materials used for their manufacture must necessarily possess high compressive strength and good resistance to wear and corrosion. Nickel alloys containing approximately 40% nickel, 40% copper, are particularly suitable for temperatures ranging up to 750°F., and the United States is especially fortunate in possessing natural ore deposits of such an alloy known commercially as Monel metal. Above 750°F. stainless steels have become increasingly important in this field and steels having a chromium content of 12 - 20% are now widely used for valves and seats. The use of stainless steel valve and seat components at elevated temperatures, however, introduced at least one serious problem, for it was found that on safety valves employing a stainless steel valve on a seat of similar material a tendency for the valve to stick occurred in practice. Investigation led to the conclusion that the high contact pressure at the seating surface combined with high local temperatures promoted the formation of a metallic bonding or pressure weld at the contact face, the long time intervals involved enabling this to occur at temperatures well below the solidus value for that particular metal. Extensive research has been applied to this problem of "cold welding," and considerable success has been achieved by the application of a chromium plate surface to the seat, the resultant combination of stainless steel in contact with chromium has been found to effectively eliminate any cold welding effects under operating conditions.

So long as adequate arrangements for draining are provided in the valve chest no serious corrosive troubles normally arise, but numerous cases of severe pitting around the valve seat have been brought to the author's attention, in some instances the threaded portion of the valve chest retaining the seat being almost entirely eaten away. The cause of the trouble in practically every case could be traced to the absence of proper draining facilities in the valve chest with the result that discharged steam condensate containing corrosive boiler compounds was able to accumulate

in the exhaust chamber. The seat itself, being stainless, generally stands up much better to such corrosive effects but a form of corrosive fatigue which attacks the grain boundaries of the metal may sometimes manifest itself in the shape of spontaneous cracking under stress. The drain connection in the valve body should, consequently, never be plugged or obstructed in any way and, moreover, the body should be properly designed to eliminate any undrained pockets in which condensate could accumulate.

High carbon steel is universally used for the springs on spring loaded safety valves although at operating temperatures above 750°F., when "crawl" effects become marked, a tungsten alloy steel is sometimes employed. This steel maintains constant strength at high ambient temperatures but suffers from a certain amount of brittleness and its use is not altogether favoured.

APPENDIX.

Design Details for Safety Valve Springs.

All springs should be proportioned strictly in accordance with the appropriate rules laid down in B.S. 759, 1950. For the benefit of readers the essential details are given below.

Maximum shear stress f_s—this may be determined from the following formulae and must not exceed a value of 80,000 lb./sq. in.

(i) Round section $f_s = K \dfrac{16\ S\ R}{\pi\ d^3}\ C$

(ii) Square section $f_s = K \dfrac{4{\cdot}8\ S\ R}{d^3}\ C$

(iii) Rectangular section $f_s = K \dfrac{(3B + 1{\cdot}8H)\ S\ R}{B^2\ H^2}\ C$

where

$K = \dfrac{4D/d - 1}{4D/d - 4} + \dfrac{0{\cdot}615}{D/d}$ In case of rectangular sections substitute B for d.

S = load in lb. at set pressure.
R = D/2 mean radius of coil (in inches).
d = diameter of round, or side of square steel (in inches).
B = breadth of wire (radial to spring axis) (in inches).
H = depth of wire (parallel to spring axis) (in inches).
D = mean diameter of coil (in inches).

C = constant = $\dfrac{L_1 + L_2}{L_1}$

L_1 = initial compression or extension of the spring to the required loading (P × A) (in inches).
where P = design pressure lb./sq. in. (set pressure).
A = loading area of valve.

L_2 = the further compression or extension of the spring to give the necessary valve lift during blow-off.

It will be readily apparent that as so many variables exist in the above formulae certain prior assumptions are imperative. A good plan is first of all to assume a mean diameter of coil (D) equal to the bore of the valve, this could be exceeded with advantage but should never be less or the lateral stiffness of the spring may be adversely affected. The next step is to calculate the maximum stress values (f_s) for different sizes of wire in order to find the minimum size wire permissible for the given conditions. This is necessarily a case of trial and error, but with experience the amount

of guesswork involved diminishes considerably and spring sizes will generally fall into a regular pattern according to size of valve and pressure range. Values of K for different ratios of D/d may be found from the graph in Fig. 25. An initial spring compression (L_1) equal to $\frac{1}{4}$ of the valve bore diameter is often adopted, but this may be increased if the spring is to be designed for use over a range of valve loadings. This initial compression must never be less than $\frac{1}{4}$ of the bore diameter.

Having determined the size of wire and the mean diameter of coil, the number of effective or free coils in the spring may be calculated from the following formulae :—

(i) Round or square wire $N = \dfrac{K\,C\,d^4}{S\,D^3}$

(ii) Rectangular wire $N = \dfrac{66\,B^3\,H^3\,K}{(B^2+H^2)\,S\,D^3}$

where N = number of effective coils.
K = compression or extension in inches at set pressure.
C = 22 for round, 30 for square steel.
d = diameter or side of square steel in 16ths of an inch.
S = load on spring in lb. at set pressure.
D = mean diameter of coil in inches.
B = breadth of wire in 16ths of an inch.
H = depth of wire in 16ths of an inch.

The space between the coils when the valve is lifted one fourth of its diameter must not be less than $\frac{1}{16}$ inch.

The number of coils and the spacing determine the free or unloaded length of the spring.* The proportion of unloaded length to external diameter of the spring must not exceed 4 to 1.

Example.—Required, spring for 3″ bore safety valve having valve lift of 0·30″. Set pressure 600 lb./sq. in.

Spring load (S) at set pressure $= 600 \times 7{\cdot}07$ lbs.
$= 4{,}240$ lbs.

Assume initial compression L_1 equal to $\frac{1}{4}$ bore diameter then constant $C = \dfrac{\cdot75 + \cdot3}{\cdot75}$

$\doteq 1{\cdot}4$

* An allowance of $1\frac{1}{2}$ - 2 coils is usually made for squaring off at the ends. These extra coils increase the overall length of the spring but they should not be added to the number of effective coils.

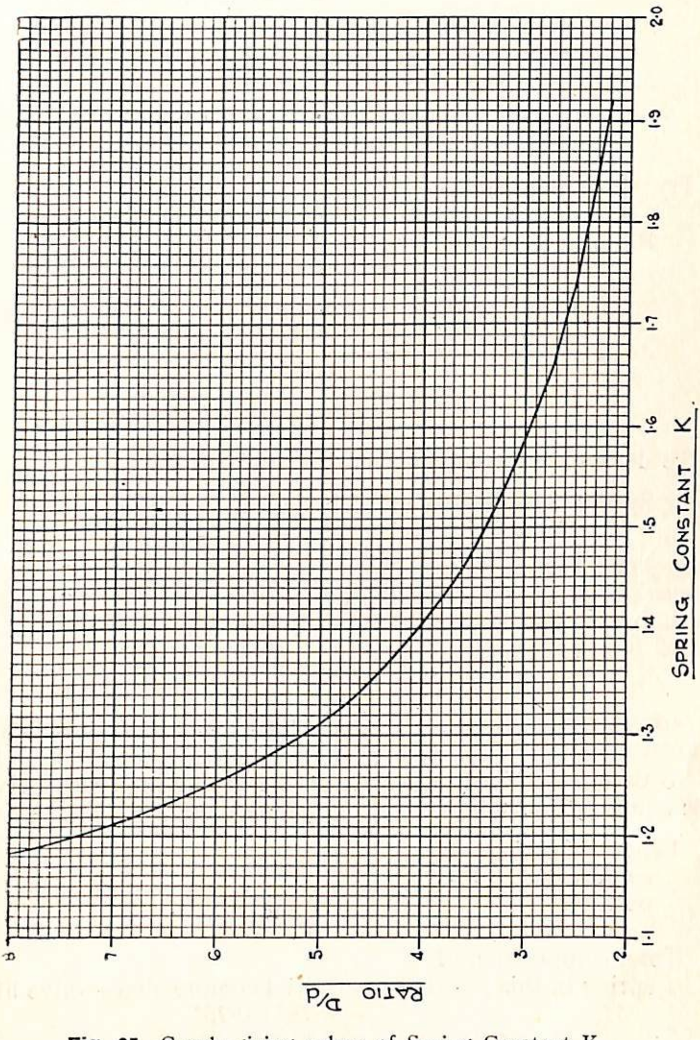

Fig. 25—Graph giving values of Spring Constant K.

Assuming a mean diameter of coil of 3″ try $\frac{7}{8}$″ square wire for maximum stress f_s :—

$$\text{Ratio } \frac{D}{d} = 3.43$$

$$\therefore \text{ K (from graph)} = 1{\cdot}490$$

$$\therefore f_s = \frac{1{\cdot}490 \times 4{\cdot}8 \times 4240 \times 1{\cdot}5 \times 1{\cdot}4}{0{\cdot}875^3}$$

$$= 95{,}100 \text{ lb./sq. in.—}\textit{Too High.}$$

Try $\frac{15}{16}$″ square wire :—

$$\text{Ratio } \frac{D}{d} = 3{\cdot}2$$

$$\therefore \text{ K} = 1{\cdot}533$$

$$\therefore f_s = \frac{1{\cdot}533 \times 4{\cdot}8 \times 4240 \times 1{\cdot}5 \times 1{\cdot}4}{0{\cdot}938^3}$$

$$= 79{,}400 \text{ lb./sq. in.—}\textit{Satisfactory.}$$

To determine the number of effective coils (N) :—

Spring compression at set pressure = 0·75″
Constant C for square wire = 30
Size of wire = $\frac{15}{16}$″
Spring load S = 4240 lbs.
Coil diameter D = 3″

$$\therefore N = \frac{0{\cdot}75 \times 30 \times 15^4}{4240 \times 3^3}$$

$$= 9{\cdot}95 \text{ coils—say 10 coils.}$$

To determine the free length of the spring let space between coils equal $\frac{1}{16}$″ when valve is lifted $\frac{1}{4}$ bore diameter then :—

Length of spring when compressed to this position = 10 ($\frac{15}{16}$″ + $\frac{1}{16}$″).
= 10″.

Total compression of spring in this position = initial compression + valve lift.
= 0·75″ + 0·75″.
= 1·5″.

∴ Uncompressed or free length of spring = 10″ + 1·5″.
= 11·5″.

Allowing $1\frac{1}{2}$ coils for squaring off at ends then :—

Total number of coils	$= 10 + 1{\cdot}5.$
	$= 11{\cdot}5.$
Total overall length of spring	$= 11{\cdot}5'' + 1{\cdot}5 \times \frac{15}{16}''.$
	$= 12{\cdot}9.''$

Summarizing final specification of spring dimensions :—

Mean diameter of coils	$= 3''.$
Size of wire	$= \frac{15}{16}''$ square.
No. of effective coils	$= 10.$
Total No. coils	$= 11{\cdot}5.$
Total uncompressed length	$= 12\frac{15}{16}''.$

Spring to compress $\frac{3}{4}''$ under load of 4240 lbs.

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